CHARACTERISTICS OF DRIFT ELIMINATORS OF AN EVAPORATIVE CONDENSER

by

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CERTIFICATE

This is to certify that the work entitled 'CHARACTERISTICS OF DRIFT ELIMINATORS OF AN EVAPORATIVE CONDENSER' has been carried out by Mr. Alok Das under our supervision and has not been submitted elsewhere for the award of a degree.

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ALOK DAS

TO MY

PARENTS AND MAMONI

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NOMENCLATURE

I	Current, Amp
n	Number of stages
$\dot{m}_a$	Mass flow rate Suction Air, kg min^{-1}
$\dot{m}_d$	Rate of drift loss, kg min^{-1}
$\dot{m}_e$	Rate of Evaporation loss, kg min^{-1}
$\dot{m}_w$	Water mass flow rate, kg min^{-1}
Δp	Total Pressure drop, mm of H_2O
Δp_1	Pressure drop in first stage, mm of H_2O
Δp_2	Pressure drop in second stage, mm of H_2O
Δp_3	Pressure drop in third stage, mm of H_2O
t	Dry Bulb Temperature (DBT), ^oC
t^*	Wet Bulb Temperature (WBT), ^oC
v	Air Velocity, m min^{-1}
V	Supply Voltage, volt
W	Specific humidity, kgw/kgda
W_1	Ambient specific humidity, kgw/kgda
W_2	Specific humidity without duct heater, kgw/kgda
W_3	Specific humidity with duct heater, kgw/kgda
θ	Inclination Angle, degree

ABSTRACT

A systematic experimental study was taken up on Drift Eliminators used in Evaporative Condensers to determine the pressure drop across them and also the drift loss. A forced draft evaporative condenser with a maximum of three removable stages of wooden drift eliminators was used in present investigation. The number of stages used during any particular experimental run was varied from one to three. Pressure drop and drift loss data were collected for various orientations of drift eliminator plates and for a range of supply voltage across the fan.

The variation of the pressure drop and the drift loss suggested an optimum angle of inclination of the plates. The optimum value depends upon the number of stages used at a time. A single stage drift eliminator with an optimum angle could be used with almost the same pressure drop and drift loss as those for two or three stages. Further a small increase in pressure loss could lead to a considerable reduction in the drift loss and a part lead operation could be possible with a reduced power consumption.

CHAPTER 1

INTRODUCTION

1.1 GENERAL BACKGROUND :

Evaporative Cooling is a process which is widely used for enhancing the efficiency of cooling of an air cooled device or space. Typical Industrial applications where this principle is used are the cooling towers in power plants or central airconditioning systems for building, Evaporative Condensers, Evaporative Coolers for process industries and desert coolers. In evaporative cooling, water is sprayed into a flowing air media and the temperature of the air is reduced considerably because of the absorbtion of latent heat by the sprayed water during the evaporation process. When this principle is applied to condensers, the cooling is done as a two fold process; firstly, the sprayed water which flows around the condenser tubes takes away some heat and secondly, the air whose temperature is brought down by the evaporating water droplets, also removes some heat from the condenser tubes.

In the past, purely water cooled type of coolers were is use whose water consumption was fairly high. In recent years however, the demand for industrial water has increased and implementing counter measures for tackling the cooling water shortage has become important.

One of the methods of achieving this is by providing the drift eliminators above the condenser tubes. The drift eliminator is typically a rectangular box fitted with plates which can be set at any desired angle to the air flow direction. By adjusting the angles of these plates with the flow direction suitably most of the water droplets can be made to fall back into the sump, thereby reducing the drift loss. The present study concerns with the performance characteristics of multistage Drift Eliminators for an evaporative condenser.

1.2 LITERATURE SURVEY :

The principle of evaporative condenser has been known for a long time. As early as 1938, Goodman discussed the advantages of using an Evaporative Condenser. Because of the wide use that such a condenser was put to in industrial applications, over the past fifty years or so, many books on Refrigeration and Airconditioning have provided a brief discussion on the evaporative condenser (Stocker, 1982; Prasad, 1985). Other common heat exchanger utilizing the principle of evaporative cooling is the cooling tower for which a lot of literature is available on its design and performance characteristics (Boelter, 1939; Uchide, 1961; Azmuner, 1962).

The relative advantages of an evaporative condenser over cooling tower are discussed in Stoecker (1982), where as a detailed discription of forced draft and induced draft evaporative condenser is given in Prasad (1985).

The thermodynamic description of an evaporative cooling process is presented in Threlkeld (1962). The state points of air through the evaporative condenser or the cooling tower and the enthalpy -temperature relationship for water and air are conveniently presented in the form of diagrams. This work also discusses the similarities and dis-similarities between evaporative condensers and cooling towers and also their relative merits and demerits. The description of an airconditioning cycle using the evaporative condenser is given in Lang (1971).

Mizushina et.al. (1967, 1968) performed experimental studies on the characteristics and design methods of evaporative coolers. They experimentally determined the heat transfer co-efficient between a process fluid and the cooling pipe, the heat transfer co-efficient between a cooling pipe and cooling water, and the overall mass transfer co-efficient between the sprayed water and the flowing air. Emperical equations for these transfer co-efficients expressing the effect of various operating factors were proposed. On the basis of these results, a method for thermal design of evaporative coolers was proposed. The design method was also illustrated by numerical examples.

The piping system with condensing coils (single and multiple) for an evaporative condenser was studied by Benner and Ramsey (1987).

Evaporative cooling with water recyclining is a viable method of making efficient use of cooling water. Another method which finds use in small scale application is the purely air cooling process. However, the lowest condenser temperature, achievable in this fashion is equal to the Dry Bulb Temperature (DBT) of the entering air. By using the evaporative cooling process, the lowest condenser temperature can be brought close to the Wet Bulb Temperature (WBT) of the cooling air. Because the Wet Bulb Temperature is always less than or equal to Dry Bulb Temperature, the refrigeration system can be operated with lower condensing temperature thereby leading to a higher COP, of the cycle. Particularly in a non-humid hot climate, the evaporative cooling can enhance the efficiency considerably.

One of the problems associated with the evaporative cooling process is the carry over of water droplets with the air flow which is formed as 'drift loss'. This drift loss is undesirably as it is a direct loss of water to the atmosphere. Particularly in places where there is a tremendous scarcity of water, this may result in a very big expense. The drift loss is also undesirable because of settling of water droplets on any electrical equipment may prove to be hazardous. Also in the case of Induced Draft Fan the water particles may intensify the corrosion of the blades thereby reducing the life of the fan. For these reasons, the thermal polution created by the transporated water droplets should be reduced.

Parallel piping configuration was considered by these Authors. They also high lighted the economics of energy utilization in evaporative condensers various energy conservation options and the design experience with indirect evaporative cooling methods were discussed by Tseng (1988).

Although evaporative condenser have been in use for a long time, till to date systematic studies have not been performed on the efficient use of Drift Eliminators for them. Details of the reduction in Drift Loss and the associated increase in power consumption while drift eliminators are used, have not been analysed. The present work attempts to investigate some of these aspects of the drift eliminators characteristics.

1.3 SCOPE OF THE PRESENT WORK :

The objectives of this research project are as follows

1. Determination of the 'Pressure Drop' and Drift Loss' for one, two, three stages of the 'Drift Eliminators' for various orientations of the drift eliminator plates in an evaporative condenser.

2. Studying the variation of the 'Pressure Drop' across the drift eliminators for various values of the air flow rates through the evaporative condenser.

The above data are taken for wooden drift eliminators and for a given water circulation rate. For the test set-up of this project, the refrigeration system capacity is also kept unchanged.

1.4 ORGANIZATION OF THE THESIS :

Chapter two describes the experimental methodology including the details of instrumentation and the method of estimating the drift loss. Chapter three describes the drift loss and pressure drop data collected during the course of this investigation. Discussion of the results including the main conclusions and suggestions for future research are provided.

CHAPTER-2

EXPERIMENTAL METHODOLOGY

2.1 TEST RIG :

The complete Test Rig consists of an evaporative condenser using a forced draft fan and drift eliminators along with a refrigeration unit which uses this condenser. The main components of the Test Rig are shown in Figure 2.1. Their description and the specifications are given below.

1. REFRIGERATION UNIT :

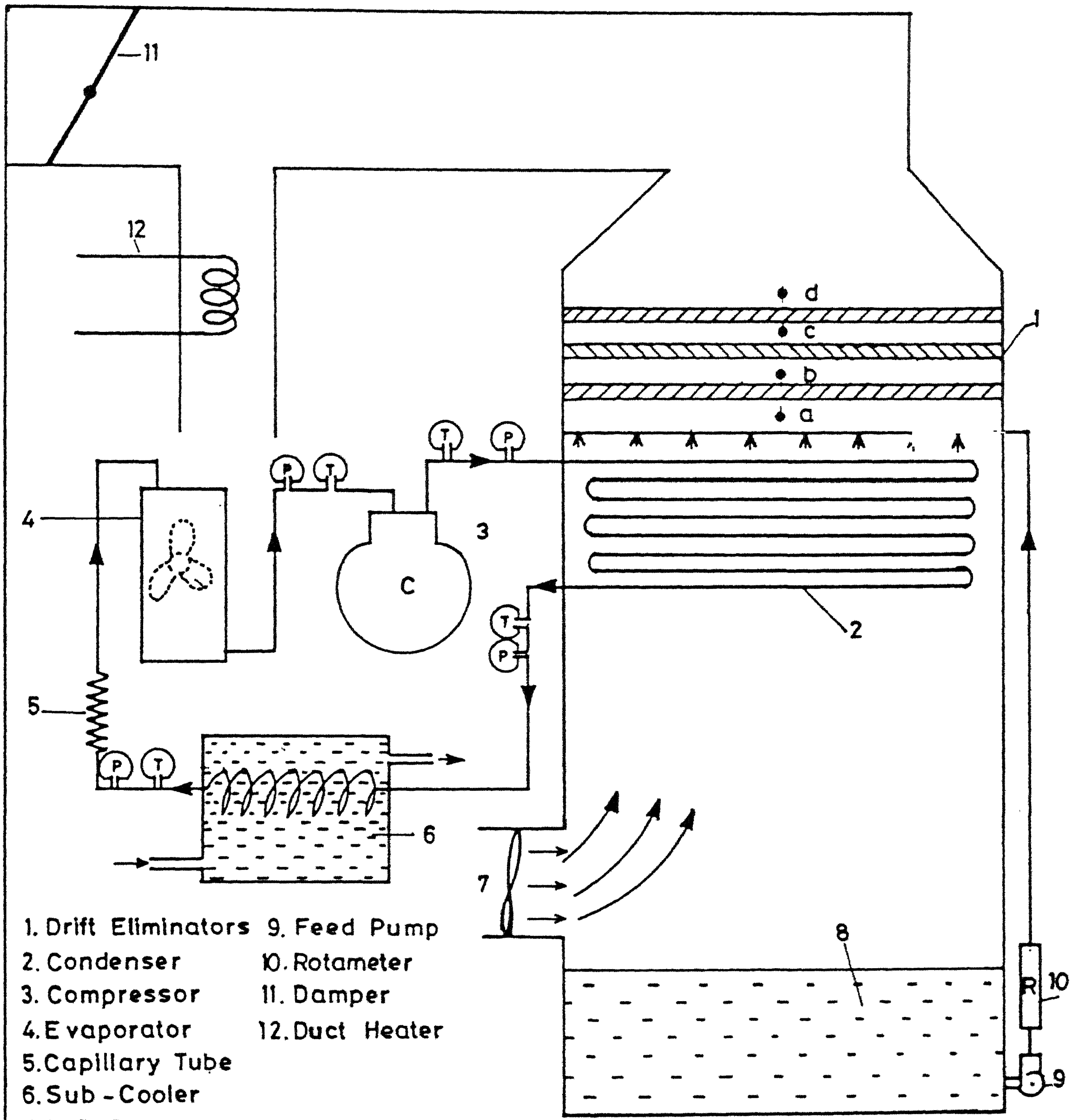
This consists of the following

a. Compressor:

It's capacity is 1.5 ton (SHRIRAM) using R-22 as the refrigerant.

Specification	:	1622
Evaporating Temperature	=	$7.2^{\circ}C$
Condensing Temperature	=	$55.0^{\circ}C$
Ambient Temperature	=	$35.0^{\circ}C$
Compressor Suction Gas Temperature	=	$35.0^{\circ}C$
Suction Pressure	=	5.425 bar or 75.95 PSIG
Discharge Pressure	=	21.467 bar or 300.54 PSIG

The Compressor draws the refrigerant from the cooling coil and increases its pressure and temperature.



1. Drift Eliminators
2. Condenser
3. Compressor
4. Evaporator
5. Capillary Tube
6. Sub - Cooler
7. F. D. Fan
8. Water Sump
9. Feed Pump
10. Rotameter
11. Damper
12. Duct Heater

Fig. 2.1 Schematic Diagram of the Test Rig.

b. Condenser:

It removes the heat picked up by the refrigerant in the cooling coil (evaporator) and condenser to a liquid state. It was designed for a capacity of 1.5 ton. This condenser removes the heat by the mechanism of simultaneous heat and mass transfer.

c. Sub-cooler :

It was provided to increase the co-efficient of performance by subcooling the liquid. It was made up of a copper tube 5/8 inch (15.88 mm) diameter and 3.01 meter long. For getting a large degree of subcooling, the provision was made to immerse this coil in a tank of water having the dimensions 0.54x0.51x0.23m.

d. Capillary Tube:

It was provided for a rated capacity of 1.5 ton as an expansion device.

e. Evaporator (Cooling Coil) :

The evaporator was 1.5 ton finned unit. A fan was installed at the back of this unit to blow ambient air for better heat transfer to the coil.
Specification of fan and motor: 0.39m dia, 930 rpm, 0.2 hp.

2. Main chamber was essentially a rectangular box 1mx0.52mx1.85m tapered at the top with a provision to connect a rectangular duct to carry the discharged air. The bottom portion of this box was used as a water sump as shown in figure 2.1.

3. Ducts:

A main duct was connected at the top of the (main) chamber to carry the discharge air. The dimensions of this duct were 0.410mx0.320mx2.26m long and was equipped with a damper to completely block the flow. A side duct of dimensions 0.18mx 0.18mx0.95m was connected to the main duct for taking the necessary measurements. A duct heater of 1.5 KW capacity was installed inside the side duct.

4. Drift Eliminators :

These were provided to reduce the drift loss. In the present case they were made of wood. Each strip of the drift eliminator could be rotated easily from 0°-180°. Maximum three stages were used during the experiments.

Specification: Area of box = 0.495mx0.95m

Thickness of each strip = 13 mm

No. of strips per box = 9

Width of the each strip = 46 mm

5. Circulating Water Pump :

It was installed to provide the spray water over the condenser coil through a set of nozzles. The nozzles were 89 holes of 2 mm dia in a pipe laid in 4 rows each of 0.85m length.

Specifications: Continuous motor, 5 hp, 2850 rpm, 4.5 amp, 230/240 volt.

6. Forced Draft Fan and Motor :

They were installed on a concrete foundation specially made for it by 4 foundation bolts.

Specifications: Fan discharge 89.9 m^3/min, static pressure 13.5 mm H_2O, 250 mm diameter, motor 0.75 hp, 1420 rpm, 5.2 amp, 230 volt.

2.2 INSTRUMENTATION USED :

Complete instrumentation was provided to measure pressures and temperatures at different points in the refrigeration system. The specifications of the gauges used are given below.

a. Compressor Suction Side :

Pressure Gauges Range	-2.142 bars to 10.71 bars or -30 to + 150 PSIG
Temperature Gauges Range	-22°C to + 50°C

b. Compressor Discharge Side:

Pressure Gauges Range	0 to 21.42 bars or 0 to 300 PSIG
Temperature Gauges Range	0 to 120°C

c. Condenser Outlet :

Pressure Gauges Range	0 to 21.42 bars or 0 to 300 PSIG
Temperature Gauges Range	0 to 120°C

d. Outlet of the Sub-Cooler :

Pressure Gauges Range	0 to 21.42 bars or 0 to 300 PSIG
Temperature Gauges Range	0 to 120°C

e. A vaccum pump was used to evacuate the refrigeration system before charging the refrigerant.

f. In order to determine the rate of water spray over the condenser coil, a Rotameter (0-5 gpm, Specific gravity 1.0) to measure the circulating water flow rate.

g. The psychrometric data of the air entering and leaving the Evaporative Condenser were measured using an ordinary psychrometer (DBT -20 to +50^{o}C, WBT -20^{o}C to +50^{o}C). To be able to measure these data for the leaving air, the damper of the main duct was closed so that all the air could flow through the side duct.

h. Velocity of the leaving air was measured using a Vane Anemometer (0 to $1x10^{5}$m, wind speed 1 to 15 m/s, OTA KEIKI SEISAKUSHO, JAPAN) at the exit of the side duct.

i. The supply voltage to the fan motor and the current drawn was measured using a digital multimeter.

Current	:	10 amp
Voltage	:	600 Volts AC
Makers	:	Simpson Pvt. Ltd, Bombay
Model	:	464

j. The pressure drop across the drift eliminators was measured using a digital Manometer.

Maximum Δp measured	:	0-199.9 mm H_2O
Specification	:	FCO 12- Micromanometer Funess Controls Ltd, Bexhill, England.

2.3 EXPERIMENTAL PROCEDURE :

Various steps of the experimental procedure are outlined below:

1. Set the drift eliminator plates at a particular angle between 0° and 90°. (The former corresponds to no flow and the latter to the maximum unobstructed flow). These drift eliminator boxes were housed in the slots meant for them as shown in Figure 2.1.
2. Switch on the refrigeration cycle.
3. Connect a Variac (Dimmerstat) in supply line to the motor of the F.D. Fan. Select a suppy voltage and start the fan. The supply voltage was varied between 180 and 230 volt A.C.
4. Switch on the circulating pump so that this water is sprayed over the condensing coil. The water flow rate was measured using the Rotameter.
5. Measure the DBT and WBT of the entering and the leaving air using a Psychrometer without and with duct heater.
6. Measure the static pressure drop between the consecutive stages of the drift eliminators using the digital manometer.
7. Measure the current drawn by the motor for each set of pressure drop data.
8. Measure the barometric pressure.

The pressure drop data were taken rather in a special manner. While the spray water was on, the pressure drops were measured by a mechanical digital manometer of an old design which although a good instrument, but was very inconvenient to use. The pressure drops were then measured without any water spray. These data for a few sets were checked with an electronic digital manometer with and without water spray. Both instruments essentially gave the same readings for the two cases. It was then decided to collect the pressure drop data using the electronic digital manometer for varying supply voltages and angles of inclination of the drift eliminator plates, but without any water spray.

2.4 ESTIMATION OF DRIFT LOSS :

As described earlier, psychrometric data were measured for the entering air and air leaving through the side duct with out and with duct heater. The damper in the main duct was kept closed.

1. MEASUREMENT OF EVAPORATION LOSS :

A simple mass balance of dry air and water over the evaporative condenser (including the side duct) is given by

$$\dot{m}_{a_1} = \dot{m}_{a_2} = \dot{m}_a \quad \text{(say)} \tag{2.1}$$

$$\dot{m}_{a_1} W_1 + \dot{m}_e + \dot{m}_d = \dot{m}_{a_2} W_2 + \dot{m}_d \tag{2.2}$$

Where

$\dot{m}_{a_1}$, $\dot{m}_{a_2}$ = mass of dry air entering and leaving the evaporative condenser.

t_1, t_2 = dry bulb temperature (DBT) of the entering and leaving air.

t_1^* , t_2^* = wet bulb temperature (WBT) of the entering and leaving air.

W_1, W_2 = specific humidity of the entering and leaving air.

$\dot{m}_e$ = rate of evaporation of water (i.e. evaporation loss)

$\dot{m}_d$ = drift loss

Equations (2.1) and (2.2) yield

$$\dot{m}_e = \dot{m}_a \; (W_2 - W_1) \qquad (2.3)$$

Both W_1 and W_2 can be easily determined knowing the dbt and wbt of the entering and leaving air. The mass flow rate of the dry air can be calculated knowing the discharge rating of the fan.

2. MEASUREMENT OF THE DRIFT LOSS :

In order to measure $\dot{m}_d$, the duct heater of the side duct was switched on so that all the drift coming past the drift eliminators could be evaporated.

Naturally now the dbt and wbt of the leaving air will be changed to new values t_3 and t_3^* corresponding to which W_3 could be easily determined. A simple mass balance over the evaporative condenser yields.

$$\dot{m}_a W_1 + \dot{m}_e + \dot{m}_d = \dot{m}_a W_3 \tag{2.4}$$

or

$$\dot{m}_e + \dot{m}_d = \dot{m}_a (W_3 - W_1) \tag{2.5}$$

Substituting from equation (2.3) into equation (2.5) yields

$$\dot{m}_d = \dot{m}_a (W_3 - W_2) \tag{2.6}$$

and

$$\frac{\dot{m}_d}{\dot{m}_e} = \frac{W_3 - W_2}{W_2 - W_1} \tag{2.7}$$

It is obvious from the above equations that both $\dot{m}_e$ and $\dot{m}_d$ can be easily determined by measuring the psychrometric data of the entering and the leaving air streams.

CHAPTER 3

RESULTS AND DISCUSSION

3.1 DRIFT LOSS AND PRESSURE DROP DATA :

In order to determine the drift loss from a varying number of drift eliminator stages and the pressure drop across them, experiments were conducted according to the procedure outlined in Section 2.3.

The angle of inclination (θ) was varied from 0° to 90° and for any given set of data, the drift eliminators were set at a particular angle. The number of stages used were three, two or one at a time. The dry bulb and Wet bulb temperatures were measured for the air entering and leaving the evaporative condenser without and with duct heater. The psychrometric data of the moist air recorded in the experiments are given in Table 3.1. As discussed in Section 2.4, the evaporative ($\dot{m}_e$) and drift ($\dot{m}_d$) losses were computed and were expressed as a percentage of the water circulation rate as shown in Table 3.2. It can be seen from this table that the evaporation loss $\dot{m}_e$ also increase either by decreasing n or by increasing θ. This is basically due to the fact that in either of the two cases (i.e. decreasing n or increasing θ), the net static pressure available for the flow increase which results in a higher volumetric discharge.

Table 3.1

Psychrometric Data of the Moist Air Entering and leaving the Evaporative Condenser.

Supply Voltage of the F.D. Fan = 230 Volts

Water Circulation Rate $\dot{m}_w$ = 20.43 kg/min.

Inclination Angle	Number of Stages	Entering Air			Discharge Air Without Duct Heater			Discharge Air With Duct Heater		
		DBT	WBT	Humidity Ratio	DBT	WBT	Humidity Ratio	DBT	WBT	Humidity Ratio
θ	n	t_1	t_1^*	W_1	t_2	t_2^*	W_2	t_3	t_3^*	W_3
degree		^oC	^oC	kgw/kgda	^oC	^oC	kgw/kgda	^oC	^oC	kgw/kgda
0	3	29.5	28.5	0.0236	33.0	29.0	0.024	35.0	29.9	0.0249
	2	29.5	28.5	0.0236	30.5	29.4	0.0255	33.0	30.3	0.0264
	1	29.5	28.5	0.0236	32.0	30.0	0.0265	34.0	31.0	0.0275
30	3	31.0	29.5	0.026	32.5	30.1	0.0265	35.1	31.5	0.0275
	2	31.0	29.5	0.026	31.5	31.0	0.028	34.5	32.0	0.030
	1	31.0	29.5	0.026	33.0	31.9	0.0285	35.0	32.9	0.0315
45	3	30.2	28.5	0.0232	30.0	29.0	0.0253	34.0	31.0	0.0278
	2	30.2	28.5	0.0232	32.0	30.1	0.0263	35.0	32.0	0.0295
	1	30.2	28.5	0.0232	32.0	30.0	0.0265	45.0	31.9	0.0295
60	3	29.0	28.0	0.0238	31.0	30.0	0.0269	35.0	32.0	0.0296
	2	29.0	28.0	0.0238	34.0	30.7	0.027	36.0	32.4	0.0305
	1	29.0	28.0	0.0238	35.0	31.2	0.0275	37.0	33.0	0.031
90	3	31.5	29.0	0.0245	33.5	31.0	0.028	36.0	33.0	0.0315
	2	31.5	29.0	0.0245	35.0	31.5	0.0285	28.0	35.5	0.0320
	1	31.5	29.0	0.0245	36.0	32.5	0.0305	38.0	34.5	0.0340

Table 3.2

Drift and Evaporation Loss

Supply Voltage of the F.D. Fan V= 230 Volts

Water Circulation Rate $\dot{m}_w$ = 20.43 kg/min.

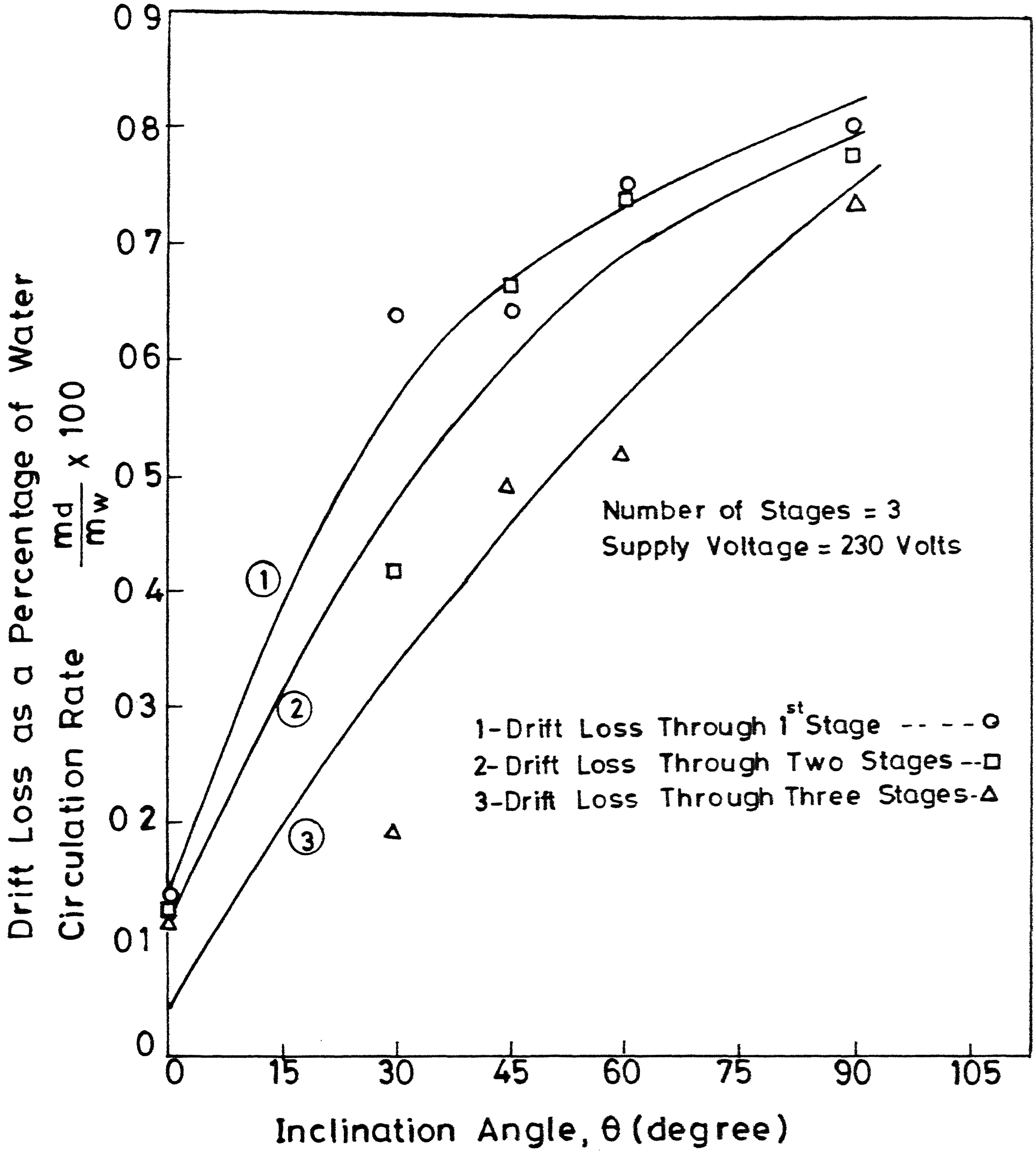
Inclination Angle θ degree	Number of Stages n	Evaporation Loss $\dot{m}_e$ kg/min	Drift Loss $\dot{m}_d$ kg/min	Mass Flow Rate of Dry Air $\dot{m}_a$ kg/min	Percent Evaporation $\dot{m}_e/\dot{m}_w$	Percent Drift Loss $\dot{m}_d / \dot{m}_w$
0	3	0.010303	0.023180	25.760	0.05043	0.1134
	2	0.050600	0.023968	26.633	0.24760	0.1173
	1	0.075260	0.028016	28.016	0.36830	0.1371
30	3	0.019738	0.039496	39.498	0.09661	0.1931
	2	0.084430	0.084430	42.215	0.41320	0.4132
	1	0.109206	0.131048	43.683	0.53450	0.6413
45	3	0.084348	0.100410	40.166	0.41280	0.4915
	2	0.133000	0.137290	42.904	0.65100	0.6720
	1	0.144506	0.131370	43.790	0.70730	0.6430
60	3	0.127226	0.106705	41.041	0.62270	0.5221
	2	0.139530	0.152620	43.606	0.68290	0.7470
	1	0.162425	0.153640	43.899	0.79500	0.7520
90	3	0.152720	0.152720	43.635	0.74754	0.7475
	2	0.182570	0.159790	45.643	0.89360	0.7818
	1	0.283700	0.165500	47.286	1.38870	0.8100

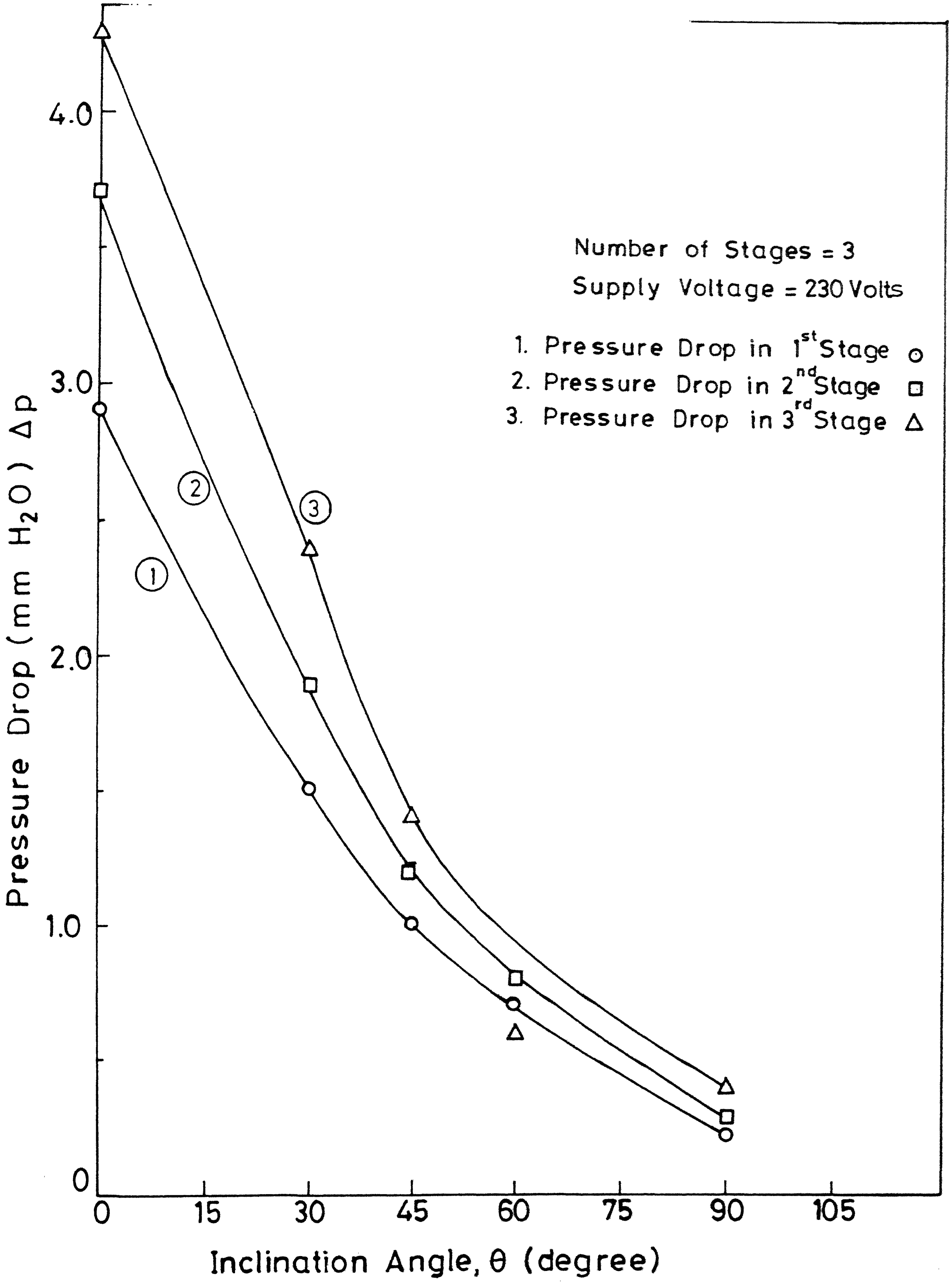
This in turn brings larger amount of air in direct contact with water resulting in a larger value of $\dot{m}_e$.

The drift loss was plotted versus θ (shown in Figure 3.1) for a supply voltage of 230 Volts A.C. curves are shown for one, two and three stages. The trend of these curves is similar. As θ increases, the drift loss increases, but it decreases with increasing number of stages. Obviously, the drift loss will be maximum for $\theta=90^{\circ}$.

The pressure drop data were recorded for θ varying between 0° and 90° and for each value of θ, the supply voltage of the forced draft fan was varied in the range of 180 to 230 Volts in order to change the speed of the fan and hence the discharge rate. Three, two or one stages were used at a time and pressure drop across each stage was recorded along with the current drawn by the motor and discharge velocity of the air through the side duct. The data recorded are shown in Table 3.3. Extracted data for the pressure drop in the individual stages of the drift eliminators alongwith total pressure drop and the drift was for a supply voltage of 230 volt across the F.D. fan are shown in Table 3.4.

The pressure drop across the various stages are plotted verus θ in Figure 3.2. It can be seen from here that as θ increases, the pressure drop across a particular set of stages decreases with minimum value corresponding to $\theta = 90^{\circ}$.


Drift Loss as a Percentage of Water Circulation Rate $\frac{m_d}{m_w} \times 100$
0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
0
15
30
45
60
75
90
105
Inclination Angle, θ (degree)
Number of Stages = 3
Supply Voltage = 230 Volts
1-Drift Loss Through 1st Stage ----○
2-Drift Loss Through Two Stages --□
3-Drift Loss Through Three Stages-△
1
2
3
Fig. 3.1



Number of Stages = 3
Supply Voltage = 230 Volts
1. Pressure Drop in 1st Stage ○
2. Pressure Drop in 2nd Stage □
3. Pressure Drop in 3rd Stage △
Pressure Drop (mm H2O) Δp
Inclination Angle, θ (degree)
0
1.0
2.0
3.0
4.0
0
15
30
45
60
75
90
105
1
2
3

Fig. 3.2

Table 3.3

Pressure Drop Data For Drift Eliminator Stages

Cross Sectional Area of the Discharge Duct = 0.0324 m^2

Inclination Angle	Supply Voltage	Number of Stages	Current Drawn	Pressure Drop				Air Velocity	Discharge Rate
θ	V	n	I	P_a	P_b	P_c	P_d	v	Q
degree	Volts		amp	mm of H_2O	mm of H_2O	mm of H_2O	mm of H_2O	m/min	m^3/min
0	180	3	3.36	13.60	11.10	10.50	9.90	630	20.412
		2	3.46	12.60	11.00	10.30	-	660	21.384
		1	3.46	13.50	10.90	-	-	720	23.328
	195	3	3.37	14.20	11.40	10.90	10.10	632	20.476
		2	3.41	14.00	11.10	10.50	-	704	22.809
		1	3.44	13.80	11.00	-	-	724	23.457
	215	3	3.45	14.30	11.60	11.00	10.30	696	22.550
		2	3.46	14.20	11.40	10.80	-	724	23.457
		1	3.49	14.20	11.40	-	-	764	24.753
	225	3	3.58	14.50	11.70	11.10	10.20	704	22.809
		2	3.57	14.40	11.60	10.80	-	725	23.490
		1	3.57	14.50	11.50	-	-	770	24.948
	230	3	3.60	14.60	11.70	11.00	10.30	708	22.939
		2	3.62	14.60	11.70	10.90	-	732	23.716
		1	3.62	14.10	11.20	-	-	770	24.948

30	180	3	3.40	12.90	11.50	10.90	10.15	790	25.596
		2	3.46	13.10	12.20	11.90	-	850	27.540
		1	3.49	13.60	11.90	-	-	1195	38.718
	195	3	3.47	12.33	11.78	11.32	10.92	880	28.512
		2	3.49	13.30	12.40	11.90	-	1172	37.972
		1	3.51	13.80	11.00	-	-	1197	38.782
	215	3	3.48	12.62	12.02	11.37	10.80	1080	34.992
		2	3.52	13.30	12.50	12.10	-	1125	36.450
		1	3.57	13.90	12.00	-	-	1190	38.556
	225	3	3.56	12.81	12.35	11.78	11.27	890	28.836
		2	3.58	13.70	12.70	12.30	-	950	30.780
		1	3.60	13.90	12.10	-	-	1175	38.070
	230	3	3.61	12.71	11.51	10.83	10.31	1076	34.862
		2	3.63	13.70	12.50	11.80	-	1150	37.260
		1	3.63	14.00	12.50	-	-	1190	38.556
45	180	3	3.41	12.50	12.20	11.60	11.40	910	29.484
		2	3.58	12.60	11.80	11.60	-	980	31.752
		1	3.48	12.80	11.20	-	-	1000	32.400
	195	3	3.44	12.30	11.80	11.50	11.20	915	29.646
		2	3.38	12.60	11.60	11.30	-	972	31.492
		1	3.45	13.00	12.50	-	-	1050	34.020
	215	3	3.48	12.40	11.70	11.30	11.10	930	30.132
		2	3.49	13.40	12.40	11.90	-	905	32.238
		1	3.50	13.60	12.00	-	-	1100	35.640
	225	3	3.57	12.70	12.30	11.50	10.40	950	30.780
		2	3.58	12.90	11.80	-	-	960	31.104
		1	3.60	13.00	12.40	-	-	1115	36.126
	230	3	3.62	13.30	12.80	12.30	11.90	1100	35.640
		2	3.64	13.70	13.10	12.50	-	1175	38.070
		1	3.65	13.60	12.60	-	-	1196	38.750

		3	3.43	12.50	12.30	12.10	11.80	915	29.646
	180	2	3.48	12.80	12.20	12.10	-	1080	34.992
		1	3.49	13.00	12.00	-	-	1197	38.782
		3	3.45	13.00	12.80	12.50	12.40	950	30.780
	195	2	3.41	13.20	12.90	12.10	-	1090	35.316
		1	3.48	13.40	11.90	-	-	1196	38.750
		3	3.50	13.30	13.10	12.80	12.70	975	31.590
60	215	2	3.52	13.60	13.20	12.90	-	1092	35.380
		1	3.57	13.70	12.90	-	-	1197	38.782
		3	3.59	13.40	13.20	13.00	12.80	977	31.654
	225	2	3.61	13.80	13.10	13.00	-	1098	35.575
		1	3.63	13.90	12.90	-	-	1196	38.750
		3	3.62	13.50	13.20	13.00	12.90	1120	36.280
	230	2	3.64	13.60	13.10	12.80	-	1190	38.556
		1	3.66	13.70	13.00	-	-	1198	38.850
		3	3.44	12.60	12.50	12.40	12.10	997	32.302
	180	2	3.61	12.70	12.50	12.40	-	1125	36.450
		1	3.50	12.70	12.50	-	-	1199	38.847
		3	3.45	12.80	12.70	12.60	12.50	1015	32.886
	195	2	3.47	12.80	12.60	12.40	-	1135	36.774
		1	3.49	12.90	12.60	-	-	1215	39.366
		3	3.48	13.10	13.00	12.90	12.80	1050	34.020
90	215	2	3.52	13.20	12.90	12.80	-	1145	37.098
		1	3.53	13.30	12.90	-	-	1260	40.824
		3	3.54	13.30	13.20	13.10	13.10	1075	34.830
	225	2	3.57	13.30	13.00	12.90	-	1144	37.065
		1	3.59	13.30	12.90	-	-	1290	41.790
		3	3.63	13.40	13.20	13.10	13.00	1195	35.316
	230	2	3.65	13.50	13.30	13.20	-	1250	40.500
		1	3.66	13.50	13.20	-	-	1295	41.958

Table 3.4

Pressure Drop and Drift for Individual Stages

Water Circulation Rate $\dot{m}_w$ = 20.43 kg/min.

Number of Stages Used n = 3

Supply Voltage	Inclination Angle	Current Drawn	Pressure Drop For Various Stages			Total Pressure Drop	Percent Drift Loss
V		I	Δp_1	Δp_2	Δp_3	Δp	$\dot{m}_d / \dot{m}_w$
Volt	degree	amp	mm H_2O	mm H_2O	mm H_2O	mm H_2O	
180	0	3.36	2.50	0.60	0.60	3.70	-
	30	3.40	1.40	0.60	0.75	2.75	-
	45	3.41	0.30	0.60	0.20	1.10	-
	60	3.43	0.20	0.20	0.30	0.70	-
	90	3.44	0.10	0.10	0.30	0.50	-
230	0	3.60	2.90	0.70	0.70	4.30	0.1134
	30	3.61	1.20	0.68	0.52	2.40	0.1931
	45	3.62	0.50	0.50	0.40	1.40	0.4915
	60	3.62	0.30	0.20	0.10	0.60	0.5221
	90	3.63	0.20	0.10	0.10	0.40	0.7475

Number of Stages Used n = 2

180	0	3.46	1.60	0.70	-	2.3	-
	30	3.56	0.90	0.30	-	1.2	-
	45	3.58	0.80	0.20	-	1.0	-
	60	3.48	0.60	0.10	-	0.7	-
	90	3.61	0.20	0.10	-	0.3	-
230	0	3.62	2.90	0.80	-	3.7	0.1173
	30	3.63	1.20	0.70	-	1.9	0.4132
	45	3.64	0.60	0.60	-	1.2	0.6720
	60	3.64	0.50	0.30	-	0.8	0.7470
	90	3.65	0.20	0.10	-	0.3	0.7818

Number of Stages Used n = 1

180	0	3.46	2.60	-	-	2.60	-
	30	3.47	1.70	-	-	1.70	-
	45	3.48	1.60	-	-	1.60	-
	60	3.49	1.00	-	-	1.00	-
	90	3.50	0.80	-	-	0.80	-
230	0	3.62	2.90	-	-	2.90	0.1371
	30	3.63	1.50	-	-	1.50	0.6413
	45	3.65	1.00	-	-	1.00	0.6430
	60	3.66	0.70	-	-	0.70	0.7520
	90	3.66	0.20	-	-	0.20	0.8100

It can also be seen from Figure 3.2 that as the number of stages increases, the pressure drop increases which in turn requires a larger amount of power.

For a particular $\theta=30^{\circ}$ (say), the reduction in the drift was for n=2 is of the order of approximately 50%, at the cost of an increase in pressure drop of the order of approximately 15% of the static pressure of the fan. Compared to n= 2, the drift loss naturally will be larger for n=1 and smaller for n=3. Pressure drop will be smaller for n=1 and larger for n=3.

The drift loss and the pressure drop were plotted versus θ for n=3, 2 and 1 as shown in Figures 3.3, 3.4 and 3.5 respectively. It can be seen from these figures that Δp vs. θ are drooping characteristics whereas drift loss vs. θ are ascending characteristics. The intersection of the two characteristic in each one of these figures indicates the particular value of θ which may be considered our optimum value. Any increase beyond this value will result in a higher drift loss, but lower pressure drop and vice versa.

It is also obvious from the above mentioned figures that as the number of stages are increased the pressure drop and the drift loss more or less are constant corresponding to optimum values of θ, which are different in each case.

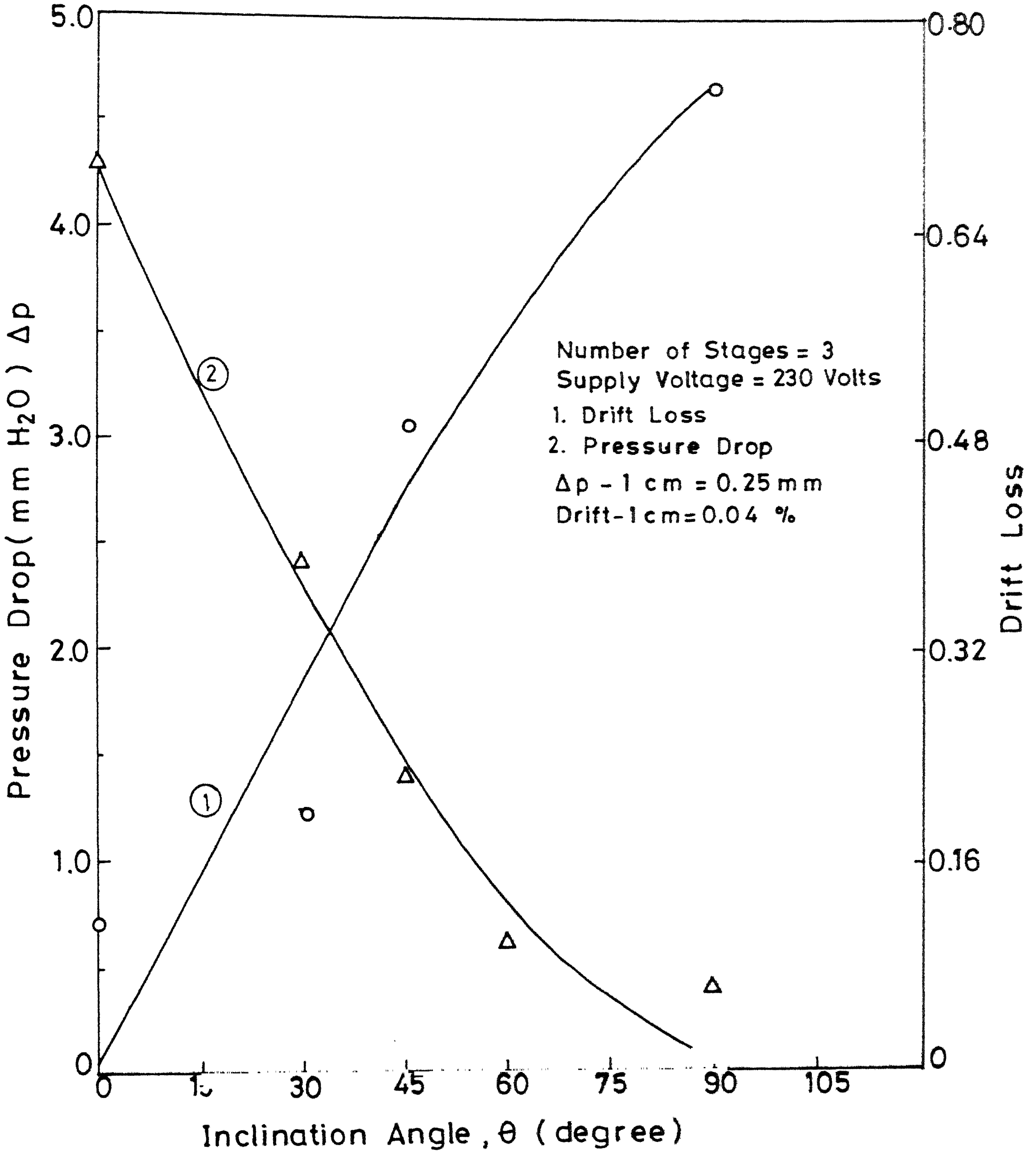

5.0
4.0
3.0
2.0
1.0
0
0.80
0.64
0.48
0.32
0.16
0
Pressure Drop(mm H_2O) Δp
Drift Loss
Number of Stages = 3
Supply Voltage = 230 Volts
1. Drift Loss
2. Pressure Drop
Δp - 1 cm = 0.25 mm
Drift-1cm=0.04 %
0
30
45
60
75
90
105
Inclination Angle , θ (degree)

Fig. 3.3

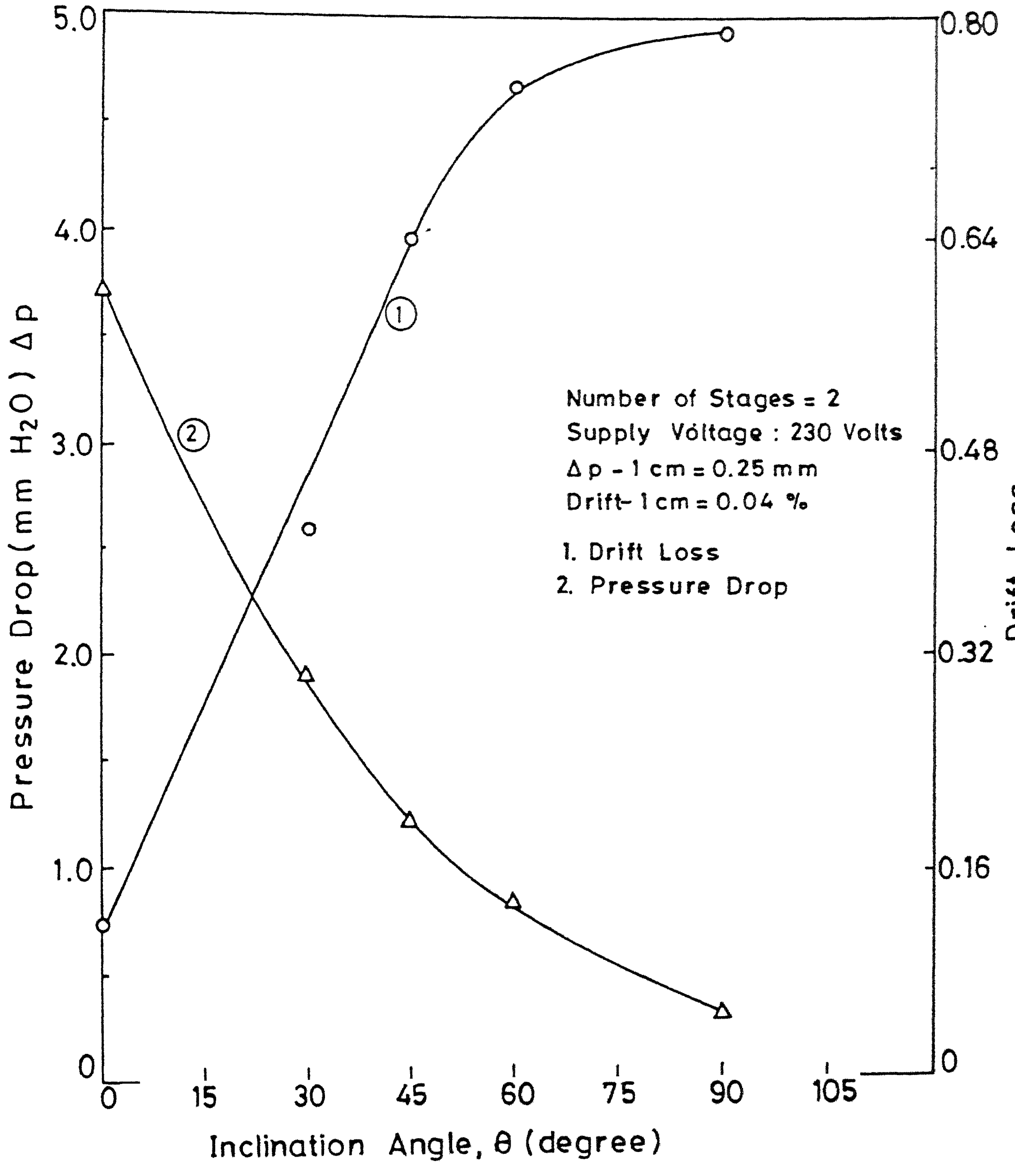
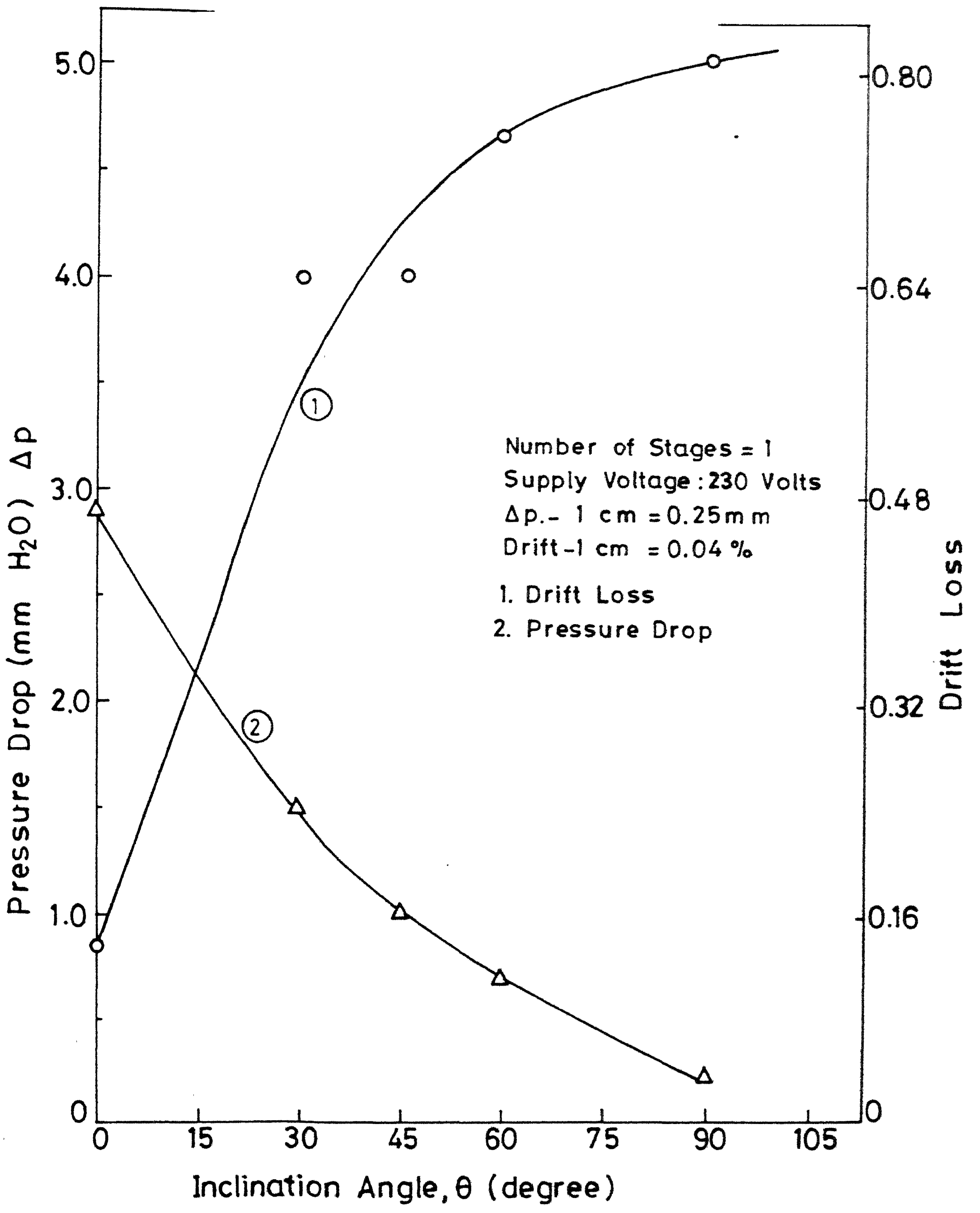

5.0
4.0
3.0
2.0
1.0
0
0.80
0.64
0.48
0.32
0.16
0
0
15
30
45
60
75
90
105
Pressure Drop (mm H_2O) Δp
Drift Loss
Inclination Angle, θ (degree)
Number of Stages = 2
Supply Voltage : 230 Volts
Δp - 1 cm = 0.25 mm
Drift- 1 cm = 0.04 %
1. Drift Loss
2. Pressure Drop
1
2

Fig. 3.4



5.0
4.0
3.0
2.0
1.0
0
0.80
0.64
0.48
0.32
0.16
0
Pressure Drop (mm H_2O) Δp
Drift Loss
1
2
Number of Stages = 1
Supply Voltage : 230 Volts
Δp.- 1 cm = 0.25mm
Drift-1 cm = 0.04 %
1. Drift Loss
2. Pressure Drop
0
15
30
45
60
75
90
105
Inclination Angle, θ (degree)

Fig. 3.5

As n increases, θ also increases which seems quite logical.

It may be concluded here that single stage drift eliminator with an optimum θ can therefore, be used for an evaporative condenser with almost the same pressure loss and the drift loss as for n=2 or 3. Increasing the number of stages will only increase the initial cost. If the number of stages were to be decided for a cooling tower, using more than one stage will be advisable from the point of view of safety and reliability particularly for an induced draft cooling tower. In this case then, the angle of inclination could be set at a larger value. It may be noted here that the above discussion regarding the optimum θ is for the forced draft fan and hot induced draft, the optimu, θ values for which may be different.

In order to study the effect of volumetric discharge rate of the fan, the supply voltage of the fan was varied in the range of 180 to 230 volts A.C. Reducing the voltage results in lower rotational speed which inturn gives a lower volumetric discharge for the same number of stages. Sample curves showing the variation of pressure drop for different stages with inclination angle are shown in Figures in 3.6 and 3.7 corresponding to supply voltage of 180 and 230 volts respectively. The trend in both figures is similar. As expected the first stage shows a larger pressure drop (curve 1) compared to second (curve 2) and second larger than the third (curve 3).

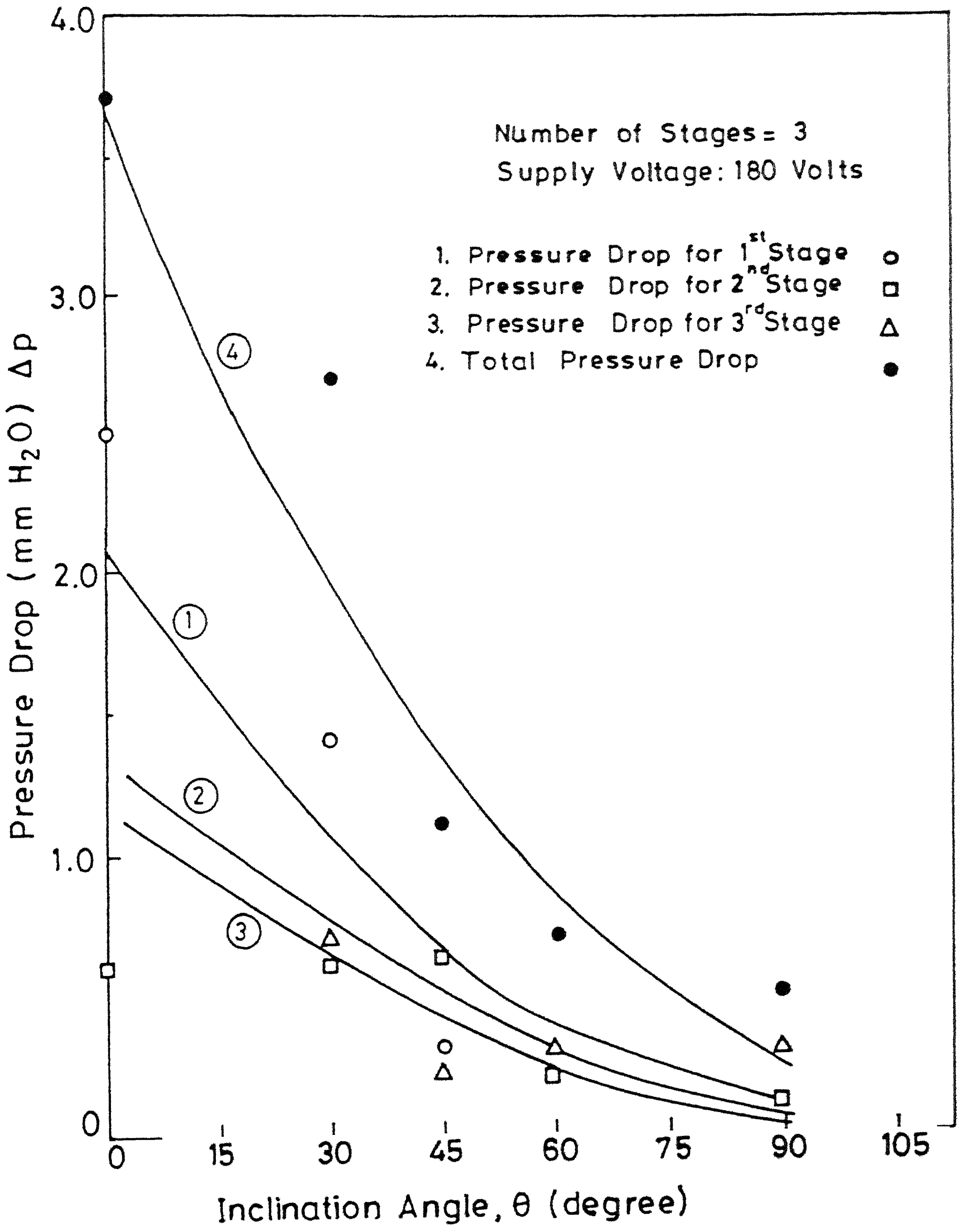

Number of Stages = 3
Supply Voltage: 180 Volts
1. Pressure Drop for 1st Stage
2. Pressure Drop for 2nd Stage
3. Pressure Drop for 3rd Stage
4. Total Pressure Drop
Pressure Drop (mm H_2O) Δp
4.0
3.0
2.0
1.0
0
0
15
30
45
60
75
90
105
Inclination Angle, θ (degree)

Fig. 3.6

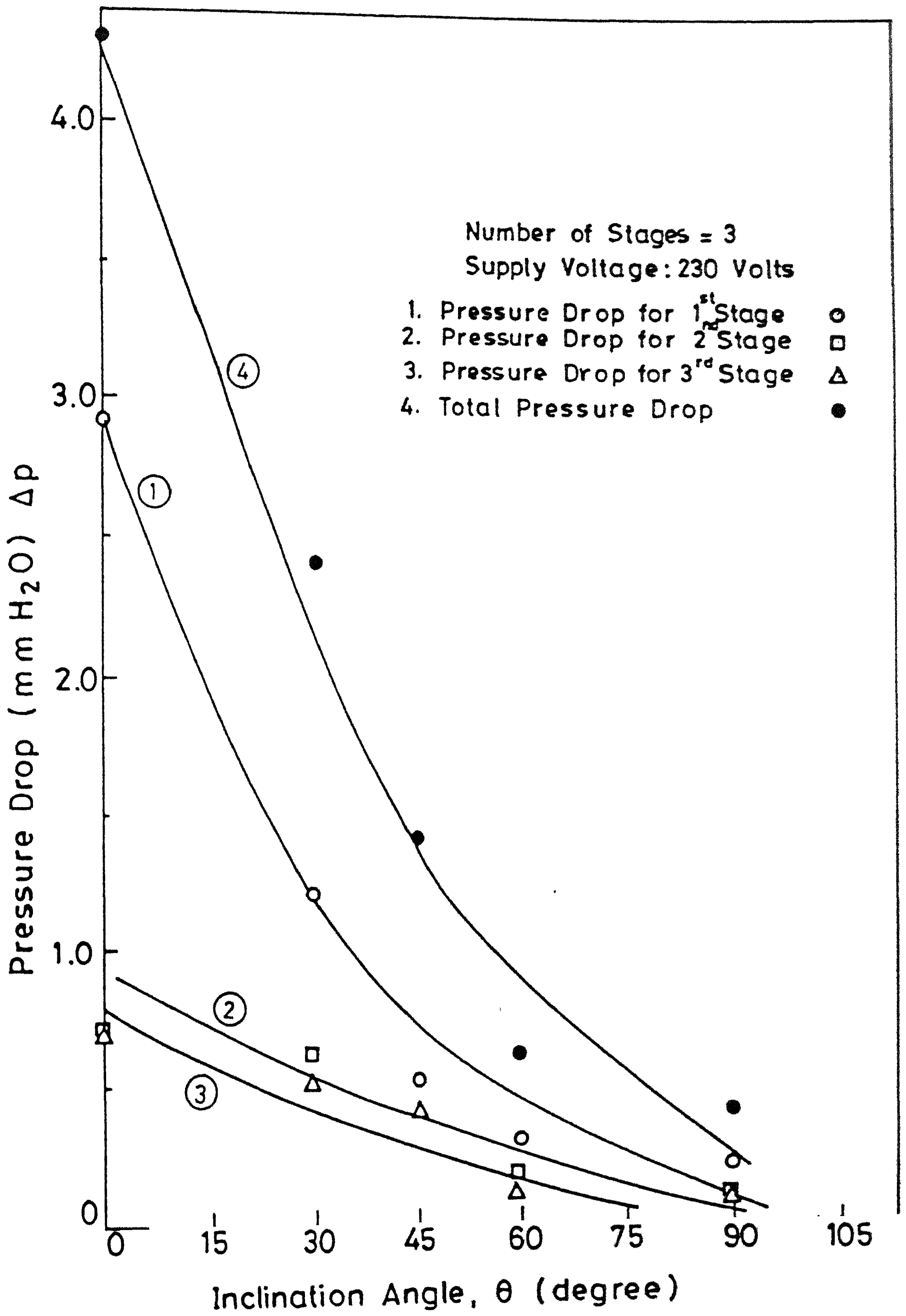

Number of Stages = 3
Supply Voltage: 230 Volts
1. Pressure Drop for 1st Stage
2. Pressure Drop for 2nd Stage
3. Pressure Drop for 3rd Stage
4. Total Pressure Drop
Pressure Drop (mm H2O) Δp
Inclination Angle, θ (degree)
4.0
3.0
2.0
1.0
0
0
15
30
45
60
75
90
105
1
2
3
4

Fig. 3.7

The total pressure drop (curve 4) for any given angle of inclination shows a larger value of Δp for 230 volt compared to that for 180 volt.

The above results can be utilized of the condenser of the Refrigeration System or the power plant were to operate at part capacity. In this case, the cooling requirement of the condenser (i.e. heat to be removed) will be smaller and one could manage with the lower flow velocity of air past the condenser and thus the lower supply voltage.

If in some application like Cooling Tower etc., it becomes essential to use more than one stage, it may be concluded from figures 3.6 and 3.7 that the power consumption by the fan will not be increased proportionately as the pressure drop in the succeeding stages decreases.

Some obvious conclusion can be drawn if the pressure drop (Δp) is plotted verus power input ($W=VI$) as shown in figure 3.8. Three average curves for $\theta=30^\circ$, 45° and 60° are shown. It can readily be seen that for constant power input Δp decreases as θ increases. If one were to maintain $\Delta p=$ constant, W would increase with increasing θ. This can be seen from the figure only for $30^\circ$and 45°, but for θ 60°, Δp is pretty well constant within the accuracy of measurement. This could be because of the cross sectional area available for the flow being sufficiently large.

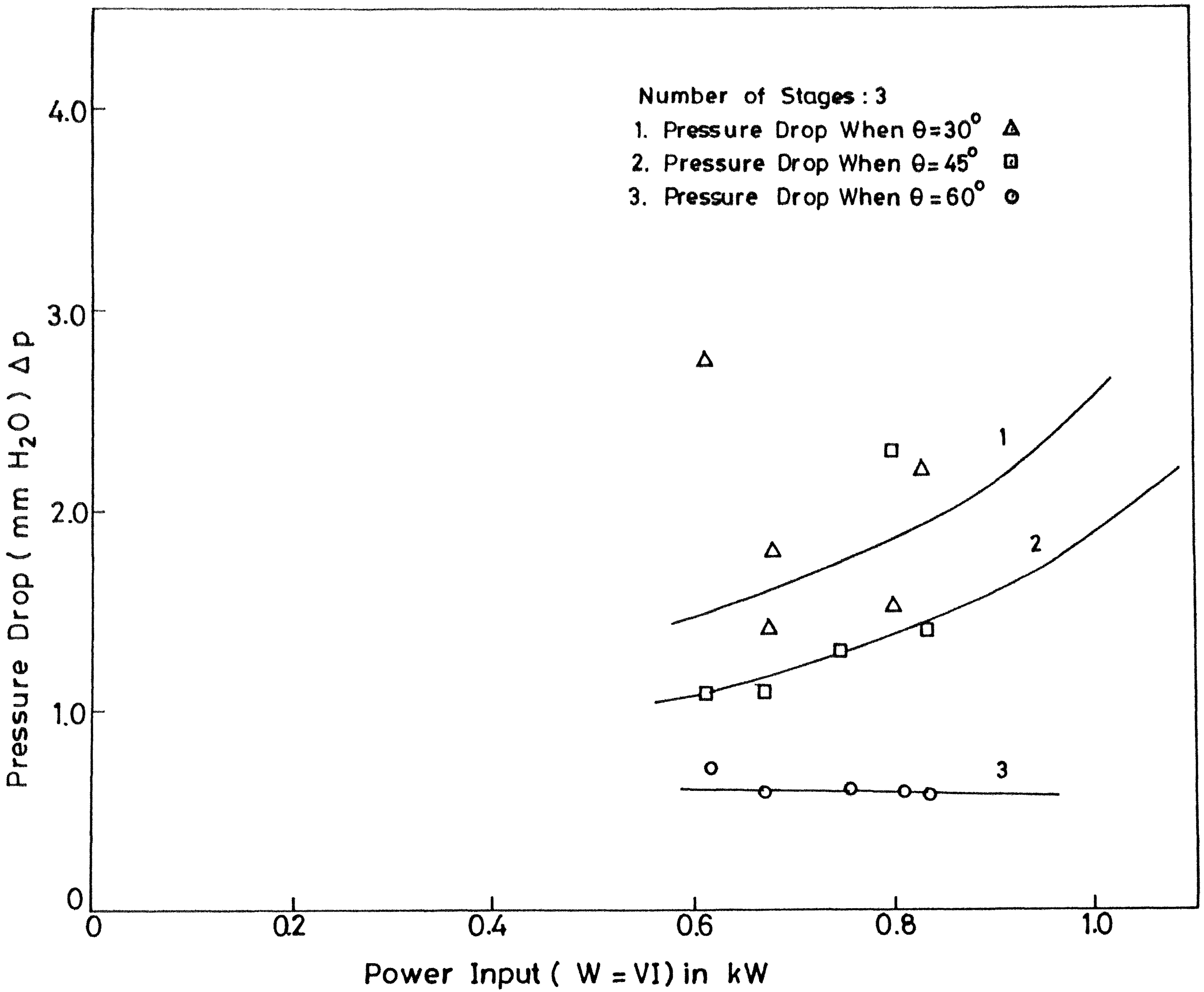

Number of Stages : 3
1. Pressure Drop When θ=30° △
2. Pressure Drop When θ=45° □
3. Pressure Drop When θ=60° ○
Pressure Drop (mm H_2O) Δp
4.0
3.0
2.0
1.0
0
0.2
0.4
0.6
0.8
1.0
1
2
3
Power Input (W = VI) in kW

Fig. 3.8

Table 3.5

Variation of Pressure Drop with Power Input

Number of Stages n = 3

Inclination Angle θ degree	Pressure Drop Δp mm H_2O	Power Input W=VI kW
30	2.75	0.612
	1.41	0.676
	1.82	0.678
	1.54	0.801
	3.20	0.828
45	1.10	0.613
	1.10	0.670
	1.30	0.748
	2.30	0.803
	1.40	0.832
60	0.70	0.617
	0.60	0.672
	0.60	0.752
	0.60	0.807
	0.60	0.832

3.2 MAIN CONCLUSIONS OF THE PRESENT WORK:

1. Inter section of the pressure drop and the drift loss characteristics yields an optimum value of the inclination angle. A single stage drift eliminator with an optimum θ can be used for an evaporative condenser with almost the same pressure drop and drift loss as for n=2 or 3.

2. Substantial reduction of drift loss is possible at the cost of a small pressure across the drift eliminator.

3. During the part load operation of the condenser, the power consumption could be reduced by operating the fan at a lower supply voltage.

3.3 SUGGESTIONS FOR FUTURE WORK:

The following additional research work can be taken up on the test rig set up during the current investigation.

1. Determination of pressure drop and drift loss for varying water circulation rates and larger range of supply voltage across the F.D. fan. Similar data should be collected for the drift eliminators made of concrete. Other geometries of the drift eliminators can also be tried.

2. Instead of using a F.D. fan similar experiments could be conducted with an Induced Draft (I.D.) fan.

It is expected that the data with both F.D. and I.D. fans put together for all the variables suggested above will be helpful in deriving more concrete conclusions regarding the type and orientation of the drift eliminator for a minimum pressure and drift losses.

In order to increase the accuracy of measurements, it would be desirable to use stabilized voltage supply and more accurate psychrometer.

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